HEAT EXCHANGER FOR AN ELECTRONIC HEAT PUMP

CROSS REFERENCE TO RELATED APPLICATIONS

[0001] This application is a Continuation-in-part of U.S. Patent Application Serial Number 10/206,731, filed July 26, 2002, which is a continuation application of U.S. Patent Application Serial Number 09/857,668, filed July 31, 2001, which is the National Phase of PCT/AU00/01220, filed October 6, 2000, which claims Foreign priority to Australia Serial Number PQ 3321, October 7, 1999.

FIELD OF THE INVENTION

[0002] This invention relates to electronic heat pumps and finned heat exchangers for transferring heat to and from such heat pumps.

[0003] For the sake of convenience, the invention will be described in relation to an electronic heat pump for a refrigeration system, but, it is to be understood that the invention is not limited thereto.

[0004] An electronic heat pump is defined herein as any heat pump or refrigerating module that directly depends upon flow of electrons and / or energy changes of electrons for its operation. This definition includes, but is not limited to, thermo-electric heat pumps and thermionic heat pumps.

BACKGROUND OF THE INVENTION

[0005] The economic viability of a refrigeration system, which is based on the principles of a electronic heat pump, is primarily dependent on the efficiency of heat exchange

between the electronic heat pump and two or more heat exchangers that collect and release the thermal load of refrigeration.

[0006] In a refrigeration system, heat can be dissipated effectively to the ambient air with the use of liquid coolants and radiators. However, the overall performance of a cooling system operating on an electronic heat pump is constrained by the heat transfer mechanism to the coolant fluid employed by the electronic heat pump.

[0007] In the prior art system disclosed in U.S. Patent No. 5,715,684, effective heat transfer is achieved by directing jets of liquid onto the face of the thermo-electric module.

[0008] In another prior art design, streams of coolant are forced to flow along a series of channels over the face of the electronic heat pump - see U.S. Patent Nos. 5,653,111 and 5,822,993.

[0009] Both of these designs offer limitations in terms of heat transfer capacity where the area available for heat dissipation to coolant is restricted to the face area of the electronic heat pump. In addition, fluid flow passages in US Patent No. 5,544,487 were made from non-conductive materials and no provision was made to incorporate additional heat flow paths to the coolant.

[0010] It is, therefore, an object of the present invention to improve the convective heat transfer between the electronic heat pump and coolant by employing a heat exchanger having a plurality of narrow channels of selected geometry.

SUMMARY OF THE INVENTION

[0011] According to one aspect of the invention there is provided a heated exchanger for use with an electronic heat pump, the heat exchanger comprising: a thermally conductive base having a first surface and a second surface, the first surface being adapted to make intimate surface contact with a surface of the electronic heat pump; a thermally conductive cover spaced from the base; a plurality of thermally conductive walls of thickness T between the base and the cover; a plurality of narrow channels defined between adjacent walls, the base and the cover through which a heat transfer liquid flows when the heat exchanger is in use; wherein each narrow channel has a width W, an inlet end and an outlet end, and; wherein the minimum number of channels/meter N is defined by the approximation: $N = 314*W-13$.

[0012] According to another aspect of the invention there is provided a heated exchanger for use with an electronic heat pump, the heat exchanger comprising: a thermally conductive base having a first surface and a second surface, the first surface being adapted to

make intimate surface contact with a surface of the electronic heat pump; a thermally conductive cover spaced from the base; a plurality of thermally conductive walls of thickness T between the base and the cover; a plurality of narrow channels defined between adjacent walls, the base and the cover through which a heat transfer liquid flows when the heat exchanger is in use; wherein each narrow channel has a width W, an inlet end and an outlet end, and; wherein the minimum thickness M(mm) of each wall is defined by the approximation: $M = 0.6*W - 0.3*W^2$.

[0013] Thus, according to the invention the preferred geometries of the channels of the heat exchanger can be described by two variables, the minimum number of channels per meter and minimum wall thickness for a given channel width. These variables control the surface area of the heat exchanger in direct contact with the fluid and the cross-sectional area available for transmission of heat by conduction up the walls.

Additional features of the present invention are set forth below.

BRIEF DESCRIPTION OF THE DRAWINGS

[0014] Fig. 1 is an exploded view of a heat pump and manifold assembly incorporating a finned heat exchanger according to one embodiment of the invention;

[0015] Fig. 2 is a cross-sectional view taken along lines ii - ii of Fig. 1 (when assembled);

[0016] Fig. 3 is an exploded view of a modified form of the heat pump and manifold assembly shown in Fig. 1;

[0017] Fig. 4 is a graph of the coefficient of performance against temperature difference for a thermo-electric heat pump;

[0018] Fig. 5 is a schematic diagram of a plurality of the heat pump and manifold assemblies shown in Fig. 1 connected in series;

[0019] Fig. 6 is a schematic diagram of a plurality of the heat pump and manifold assemblies shown in Fig. 1 connected in parallel;

[0020] Fig. 7 is a schematic diagram of a refrigeration system incorporating the heat pump and manifold assembly of Fig. 1;

[0021] Fig. 8 is a cross-sectional view of fins of a heat exchanger according to another embodiment of the invention;

[0022] Fig. 9 is a cross-sectional view of fins of a heat exchanger according to another embodiment of the invention;

[0023] Fig. 10 is an exploded view of a heat pump and manifold assembly incorporating two heat pumps according to another embodiment of the invention;

[0024] Fig. 11 is a perspective view of the heat pump and manifold assembly shown in Fig. 10;

[0025] Fig. 12 is a perspective view of one of the heat exchanger fin arrays shown in Fig. 10,

[0026] Fig. 13 is an enlarged view of portion of the heat exchanger fin arrays in Fig. 12;

[0027] Fig. 14 is a perspective view of the other fin array shown in Fig. 10;

[0028] Fig. 15 is an enlarged view of part of the fin array shown in Fig. 14;

[0029] Fig. 16 is a graph of the Nusselt number against Reynolds Number for fully developed flow in a duct;

[0030] Fig. 17 is a graphical representation of coolant temperature profiles inside a channel of the finned heat exchanger shown in Fig. 1;

[0031] Fig. 18 is a graphical representation of coolant temperature profiles inside the passageway of a prior art manifold;

[0032] Fig. 19 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:10;

[0033] Fig. 20 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:6;

[0034] Fig. 21 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:4;

[0035] Fig. 22 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:3;

[0036] Fig. 23 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:2;

[0037] Fig. 24 is a graphical representation of coolant temperature profiles inside a micro channel having an aspect ratio of 1:1;

[0038] Fig. 25 is a graphical representation of the relationship between thermal resistance and channel height with an exemplary channel width ranging from 0.05mm to 0.70mm with the wall thickness kept equal to the channel width;

[0039] Fig. 26 is a graph of thermal resistance versus the number of channels per meter for several different widths of channel;

[0040] Fig. 27 is a graph of the minimum number of channels per meter versus the channel width;

[0041] Fig. 28 is a graph of the thermal resistance versus wall thickness for a number of different widths of channel;

[0042] Fig. 29 is a graph of minimum wall thickness versus channel width;

[0043] Fig. 30 is a schematic representation of an exemplary heat exchanger showing the dimensions used in the description;

[0044] Fig. 31 is a cross-sectional view of an extruded heat exchanger according to one embodiment of the invention;

[0045] Fig. 32 is a cross-sectional view of a heat exchanger according to another embodiment of the invention; and

[0046] Fig. 33 is a cross-sectional view of a further embodiment of the invention.

DETAILED DESCRIPTION OF ILLUSTRATIVE EMBODIMENTS

[0047] The heat transfer system 10 shown in Figs 1 and 2 includes an electronic heat pump 11 having, in this instance, an upper cold side 12 and a lower hot side 13, a cold side finned heat exchanger 14 including a cold side backing plate 15 and a cold side manifold 16. In other embodiments, the electronic heat pump 11 may be substituted with other thermoelectric or thermionic modules. On the hot side of the electronic heat pump 11 there is a hot side finned heat exchanger 17 including a hot side backing plate 18 and a hot side manifold 19.

[0048] The finned heat exchangers 14 and 17 each consist of a flat base plate 15 integral with or joined to a plurality of parallel equally spaced fins 21.

[0049] In order for the system to function, a liquid coolant is passed through the channels between the fins 21 of the heat exchanger 17. Heat is then transferred away from the "hot side" of the thermoelectric module by conduction through the coolant in the heat exchanger channels and from the surface of the heat exchanger, conduction through the heat exchanger 17 and through the solder or other jointing compound fixing the heat exchanger 17 to the adjacent surface of the electronic heat pump 11. Heat is transferred through the electronic heat pump 11 in its normal manner. The second heat exchanger 14 may or may not be attached to the "cold

side" of the electronic heat pump 11 and operates in a similar fashion to the heat exchanger 17 on the "hot side" but with the direction of heat flow reversed.

[0050] The respective orientation of the cold side and hot side are controlled by the electrical polarity of the electronic heat pump 11.

[0051] The dimensions of the system are based on the dimensions of the electronic heat pump 11, which is determined by its manufacturer.

[0052] In one embodiment, the heat exchanger 14, 17 consists of a flat base plate 15, 18 joined to a plurality of axially aligned, equally spaced fins, enclosed by a flat plate (e.g. 20) across the top of the fins. In another embodiment, the flat plate across the top of the fins is integral with the fins, forming channels surrounded by homogenous parent metal. The number of fins, the dimensions of the fins, the dimensions of the space between the fins are optimized by numerical analysis of flow and heat transfer to ensure the most efficient convection for a minimum of flow resistance. The cross-sectional shape of the fins may be further optimized from the simple rectangular shape to a more complex shape such as a trapezium to further heat transfer or to facilitate manufacture.

[0053] The surface of the base plate 15, 18 of the heat exchanger 14, 17 in contact with the electronic heat pump 11 is manufactured to sufficient flatness to ensure good thermal contact with the electronic heat pump 11. The heat exchanger 14, 17 is made of a material with high thermal conductivity, and is mechanically robust and resistant to corrosive damage by the coolant.

[0054] Each manifold 16 and 19 has the following functions: (a) an enclosure to receive and discharge the coolant, via ports 100, from an attached pipe, and (b) a flow distributor to evenly distribute flow of coolant between the adjacent fins of the heat exchanger 14, 17. To serve function (a), each manifold 16, 19 is fitted with an entry and exit port 100 for fluid, the entry and exit ports are located at opposite ends of a diagonal that is drawn across the rectangular cross section of the cover. The purpose of this orientation is to ensure even distribution of flow

to the fins 21, according to an earlier established principle as discussed in U.S. Patent No. 5,653,111.

[0055] Adjacent to the exit and entry ports 100, there is a cavity 101 running from the port 100 to at least the furthest fin 21. The purpose of the cavity 101 is to ensure an even distribution of flow from the port 100 to the fins 21 of the heat exchanger 14, 17. Each manifold 16, 19 may be fitted with an equally spaced series of bolt-holes 102 running around the periphery of the cover. This provision allows bolts and nuts to impose the clamping force.

[0056] As shown in Fig. 2, the electronic heat pump 11 is sandwiched between the two heat exchangers. In the instance of a Peltier cell, the ceramic exterior faces 110, 111 are in close contact with the base plates 15, 18 of the heat exchangers. The base plates 15, 18 are restrained by their side edges soldered to a metallized surface on the ceramic faces 110, 111 and may be sealed against the interior surface 112 of the manifolds 16, 19. O-ring seals 113 may be used to prevent leakage of fluid from the channels 101 into the central area 114 containing the electronic heat pump 11. As further illustrated in Fig. 2, the ports 100 lead into channels 101 which extend at least the full length of the array of fins 21. The distal edges of the fins 21 or alternatively, the plate or surface 20 which encloses them is in contact with the interior surface of the manifold 16, 19.

[0057] Fig. 2 illustrates two distinct styles of heat exchanger fabrication. The upper or cold side heat exchanger comprises an array of fins 21 and the base plate 15. In this example, the array of fins and channels 21 include a covering plate 20 which may be integral with the fins or soldered onto the array of fins. It is this covering plate 20 which is in contact with and sealed against the manifold 16 so that fluid flow between the channels 101 occurs only through the array of fins 21. Where manufacturing tolerances can be controlled, and as shown in the lower half of Fig. 2, the array of fins 21 may be open ended, with the distal tips of the fins 21 contacting and sealing against the floor of the manifold 19. A third variation is depicted in Fig. 3.

[0058] Fig. 3 illustrates a resilient polymeric sheet 120 interposed between one or both heat exchangers and their respective manifolds 16, 19. These polymeric or soft metal sheets 120 may be used to ensure a proper resilient seal between an array of fins 21 and its manifold when the manifolds are joined together. If effect, the sheets 120 are capable of taking up manufacturing tolerances, or in the case where open ended fins are used (as shown in Fig. 3) actually serve to seal the channels between fins 21 against the inner surface of the manifold.

[0059] The efficiency of a thermoelectric heat pump is critically dependent on the temperature difference between the hot side and the cold side. Fig. 4 shows a graph of COP (coefficient of performance) versus del T for a typical thermoelectric module (Frost 76S from Kryotherm).

[0060] Figs. 5 and 6 show a series and a parallel arrangement of heat exchanger 'units' 10 to obtain a larger refrigerating power than can be achieved with a single heat exchanger and enclosed electronic heat pump. Fig. 5 illustrates a series arrangement of devices 10 of the type depicted in Fig. 1. It would be appreciated that by fluidly connecting adjacent devices 10 in a counter-current arrangement can result in the ability to accommodate greater thermal loads for a given rate of fluid flow. In this example, the hot side of the device 10 is connected to the hot side of an adjacent device, the flows of hot and cold liquids traveling in opposite directions as illustrated. Fig. 6 illustrates the parallel connection of two pairs of devices 10, each pair operating in series. Again, the flows of hot and cold liquids are traveling in opposite directions to maximize thermal efficiency. The hot side fluid flows 130 are depicted as a solid line while the cold side fluid flows are illustrated with a dash line 131.

[0061] Fig. 7 illustrates a schematic system diagram illustrating an application of the device 10 of the present invention. In this example, a cold side heat secondary exchanger 150 is located within a refrigerated space 151. A small fan 152 circulates the air within the refrigerated space in an attempt to achieve thermal equilibrium. The cold side secondary heat exchanger 150 is supplied with cold fluid from the electronic heat pump 10 by a pump 153. The output of the

electronic heat pump's hot side manifold is delivered to a secondary hot side fan assisted heat exchanger 154, circulation between the secondary heat exchanger 154 and the heat pump 10 being accomplished by a second pump 155.

[0062] Fig. 8 illustrates an array of fins 161 which may be used in place of the rectangular fins depicted in, for example, Figs 1 and 3. These fins 161 are tapered and include longitudinal grooves 162 which serve to increase the surface area interface between the fins 161 and the channels 160. In this example, the side surfaces of each fin are provided with a pair of "V" shaped grooves which promote heat transfer between the fin 161 and the channel 160. The same effect may be achieved by other forms of convolution of the fins surface or by roughening the surface of the fin.

[0063] Fig. 9 illustrates an alternate embodiment of an array of fins wherein the individual fins are replaced by a corrugated metal sheet 170 which is interposed between a pair of parallel sheets or plates 171, 172.

[0064] As shown in Fig. 10, two or more electronic heat pumps 11 may be stacked into a single working module 180. In this example, the cold sides 12 of a pair of heat exchangers 11 are arranged in a facing relationship and separated by a single finned heat exchanger 181. Each hot side 13 of the pair of electronic heat exchangers is associated with its own manifold and heat exchanger 182.

[0065] As shown in Fig. 11, liquid enters the upper and lower manifold entry ports 190 and exits through the hot side ports of the upper and lower manifolds 191. The central manifold and heat exchanger 192 circulates fluid past the cold sides of both of the heat pumps within the module 180.

[0066] Fig. 12 illustrates an array of fins 200. Each fin 201 is generally rectangular in cross section. Each pair of adjacent fins defines a microchannel there between. As shown in Fig. 13, the ends 202 of each fin 201 may be provided with a step 203 for the purpose of facilitating attachment to the manifold.

[0067] Fig. 14 illustrates the type of fin array which is required for the central manifold 181 depicted in Figs. 10 and 11. As shown in Fig. 15, the array comprises a central web 204 which has similarly configured fins 205 directed outwardly from both its upper and lower surfaces.

[0068] The efficiency of the heat pump will be enhanced significantly if the same amount of heat can be pumped from the hot or cold side at a lower temperature difference between the surface of the thermoelectric module and the liquid passing through the heat exchanger. Since heat flow is equal to h_c x Area x d T (where h_c is the heat transfer coefficient), a relatively simple way to reduce d T is to increase Area. The design of the heat exchanger with multiple fins achieves this aim and leads directly to greater heat pump efficiency.

[0069] Further, however, there are several other important benefits that the narrow microchannels design confers. It has been found through recent research into the cooling of high heat load computer chips that the usage of microchannels leads to unexpectedly high heat transfer coefficients. The reasons are not yet clear but are believed to include the increased impact of surface tension and electric potential effects which lead to earlier transitions from laminar to turbulent flow. The effects of natural surface roughness are also magnified in microchannel flow and can contribute to the high heat transfer coefficients.

[0070] When applied to cooling computer chips, very high heat loads are encountered. Heat fluxes of 75 W/cm^2 are now being achieved. These high heat fluxes result in high dT's across conventional heat exchangers. Thus the efficiency of thermoelectric devices are significantly reduced when used in conjunction with conventional heat exchangers (refer Fig.4). The heat exchanger design of the present invention exploits the high heat transfer coefficients possible with microchannels and applies the benefit to achieve relatively low d T's. These conditions are ideal for thermoelectric heat pumps and lead to significantly enhanced efficiencies.

[0071] Heat transfer in laminar flow is by conduction rather than by convection as is the case in turbulent flow. Because most liquids, including water, have low thermal conductivities this means that heat transfer coefficients are relatively low. The flow in the heat exchangers of this design is in the laminar region and particular attention must then be paid to heat transfer coefficients because of the deleterious effects of high temperature differentials on the thermoelectric module.

[0072] A benefit which is exploited in the design is the known feature that the h_c in developing laminar flow is significantly higher than in fully developed laminar flow. The length of channels is controlled to a significant degree by the physical size of the thermoelectric module, typically 40 mm square, and the dimensions of the channels have been optimized within these restrictions so that flow exists predominantly in the developing region.

[0073] It is possible to increase the rate of convective heat transfer, without using a finned heat exchanger, by increasing the flow speed of the coolant over the exterior of the electronic heat pump when the flow is in the turbulent region. The heat transfer coefficient is approximately proportional to flow rate when this occurs.

[0074] As shown in Fig. 16, however, when the flow is laminar, according to the Nusselt equation from the theory of heat transfer in laminar flow, the heat transfer coefficient is related to flow velocity to only the power of 0.3. In other words, increasing flow speed has very little beneficial effect on the heat transfer coefficient. In a laminar flow pump, power is proportional to the square of the flow rate and therefore if this strategy is adopted, it will have a negative impact on overall system efficiency, i.e. the total electric power (including thermoelectric module, pumps and fans) required to pump a given amount of heat will rise.

[0075] The adoption of a finned heat exchanger with its increased surface area and improved heat transfer coefficients due to the effect of the microchannels enables more efficient optimization of the ancillary power consumption of the pumps and fans.

[0076] Heat flux from the walls of the channel into the liquid coolant is optimized when all parts of the channel surface are at a uniform temperature. The design of the heat exchanger is such that this is achieved through careful consideration of fin height as well as spacing. The length of the fin is critical because thermal resistance is proportional to fin length. The narrow width of the channel eliminates the situation where the bulk of the fluid passes straight through a heat exchanger with the heat transfer restricted to a relatively thin film of fluid at the surface.

[0077] Fig. 17 shows temperature contours within a micro channel of one embodiment of a finned conductive heat exchanger having an aspect (i.e. width to height) ratio of 1:3.5 on the hot side of a heat pump, the heat flux being 40,000 W/m^2, inlet fluid temperature 27°C, flow rate 1 l/min with pure water coolant. These temperature gradients show minor variation (2.4°C) across the fluid, indicating that all of the fluid is involved in the heat transfer process with little bypass.

[0078] Fig. 18 shows temperature contours within a channel of a finned insulating heat exchanger having an aspect ratio of 1:3.4 on the hot side of a heat pump, the heat flux being 40,000 W/m^2, inlet fluid temperature 27°C, flow rate 1 l/min with pure water coolant.

[0079] The critical feature of the temperature profile is the difference in temperature between the fluid close to the heated surface and the bulk of the fluid. It can be seen that this difference is significantly less for the heat exchanger shown in Fig. 17 than for the earlier design involving plastic fins or partitions shown in Fig. 18 which has a temperature gradient of 30.7°C. This indicates that the heat exchanger has largely solved the problem of the earlier design where the bulk of the coolant remained effectively unheated during its passage through the heat exchanger.

[0080] The heat dissipation capability of the narrow channel heat exchanger is primarily dependent on the conduction of heat along the walls of the channel and the convective heat transfer in the fluid at the channel walls. The combination of these two aspects determine the overall thermal resistance of the heat transfer process within the heat exchanger. Increased

channel wall thickness and enhanced convective mechanism resulting from higher fluid velocities act favorably to reduce the overall thermal resistance in the heat exchanger.

[0081] Using a computational heat and fluid flow model, the heat transfer performance of the narrow channel heat exchanger is evaluated and optimized to obtain the most effective flow arrangement. For a given fluid mass flow rate and a fixed external heat flux applied to the top surface of the channel, the variation of fluid temperature contours with channel aspect ratio is illustrated in Figs. 19 to 24.

[0082] It is evident that, as the channel aspect ratio increases (narrow channel), heat tends to penetrate deeper into the fluid passage reducing the difference between the highest and the lowest temperatures indicated in the fluid. Consequently, the fluid temperature distribution becomes more uniform in these channels. Thus, the narrow channels tend to exhibit a lower thermal resistance (or a higher thermal conductance) for heat flow to the fluid than the equivalent channels of small aspect ratios. The mechanisms of convective heat transfer enhancement in narrow channels and the extended area available for heat dissipation are the primary factors that contribute to this behavior. High thermal conductivity of channel wall also effectively helps to achieve further improvements in heat transfer performance.

[0083] While the heat transfer capability improves with the increased aspect ratio, higher fluid pumping power requirements in narrow channels determine the upper limit of the useable range of aspect ratio for these channels. The range of aspect ratios found to be useful range from 4:1 to 50:1. When applied to a typical thermoelectric module which has surface dimensions of 40mm x 40mm the number of channels may range from a minimum of 10 up to a maximum of 200.

[0084] In one embodiment, a thermally conductive base plate is integrated with the fins to ensure minimal thermal resistance to heat flow. This base plate could act as the wall of an electronic heat pump, replacing the low conductivity ceramic presently used.

[0085] Careful control of thermal contact resistance between heat exchanger base plate and electronic heat pump is critical to achieving high thermodynamic efficiency of the system. The extremely low thermal conductivity of air (approximately 0.03 W/m*K) causes a high thermal impedance to be generated by any gap exceeding approximately 5 micrometers thickness. Consequently, both contacting surfaces of the heat pump and the heat exchanger must be flat to within approximately 1 micrometers tolerance to ensure a satisfactorily small contact gap. In low-cost manufacturing, such a small tolerance may be difficult to achieve so a solder joint may become necessary. The solder should have the highest practical level of thermal conductivity and a low melting point to facilitate the joining of the heat exchanger to the surface of the electronic heat pump, without damage to the latter. Alternatively, a flexible film can be used to replace the conventional rigid ceramic plate used as the external surface of a thermoelectric module. This film must be thermally conductive, electrically insulating and flexible to compensate for surface irregularities. Flexibility also enables the film to absorb shear stresses brought about by materials with different thermal expansion coefficients on either side of the film.

[0086] The overall size of the heat exchanger is not limited to the surface area of the electronic heat pump. It can be made larger and because it is of medium to high conductivity material there will be minimal thermal resistance to the flow of heat. This enables an even greater expansion of the surface area for heat exchange to a liquid coolant through channels.

[0087] Other high conductivity devices, such as heat pipes, can be used in conjunction with the heat exchanger in order to enlarge the potential contact area or to transport the heat load to a more convenient location for mounting of the heat exchanger.

[0088] In order to appreciate the enhanced mechanism of heat transfer provided by the invention for high heat flux thermoelectric cooling applications, it is appropriate to review the development of heat transfer techniques.

[0089] In cooling of electronic equipment, traditional heat transfer mechanisms such as natural convection, forced convection and boiling have been effectively applied and tested. In the past decade, requirement for operating heat flux levels of these devices has been steadily increasing from around 50 W/cm^2 to 100 W/cm^2. Even with various enhancement methods, conventional heat transfer equipment is inadequate for most of these applications owing to their poor thermal characteristics and large physical size. The quest for miniaturisation in modern devices has crated an urgent need for development of high heat flux modules and improved understanding of heat transfer phenomena.

[0090] The prior art includes many heat transfer mechanisms that generally yield significantly high levels of heat fluxes. Some such flow arrangements with inherently high rates of heat transfer are jet impingement cooling, interrupted jet cooling and heat transfer in very narrow passages or microchannels.

[0091] In jet cooling techniques, the thermal and hydrodynamic boundary layers associated with the flow are continuously changed causing a reduction in thermal resistance at the liquid-wall interface. Hence, the heat dissipation to the fluid is improved. However, due to high jet flow velocity requirements and wetting of surfaces, applications are limited to specific cases of heat transfer situations. In a microchannel heat exchanger, a cooling liquid is forced through narrow channels (width of the order of 0.05 to 5mm) built in a plate attached to an electronic device to carry away the heat generated during its operation. Through experimental methods, it has been established that, the heat transfer coefficients in microchannel flow tends to be about 60 times higher than those of conventional macroscale flow passages. Microchannel heat transfer is considered to have great potential for providing high rates of cooling necessary for modern instruments with high powered circuitry in applications such as Micro-Electric-Mechanical-Systems, high-speed computers, biomedical diagnostic probes, lasers and precision manufacturing.

[0092] Various studies indicate that the microchannel flow and heat transfer phenomena cannot be explained by conventional theories of transport mechanisms. For instance, the transition from laminar flow to turbulent flow starts much earlier (e.g., from Re = 300); the correlations between the friction factor and the Reynolds number for microchannel flow are very different from that in classical theory of fluid mechanics; the apparent viscosity and the friction factor of a liquid flowing through a microchannel may be several times higher than that in the conventional theories. These special characteristics of flows and heat transfer in microchannels are the results of micron-scale channel size and, the interfacial electrokinetic and surface roughness effects near the solid-liquid interface. High convective heat flux rates achievable in microchannel flow is attributed to these vastly different flow phenomena that occur in narrow passages.

[0093] High rate of heat flux encountered in microchannels allow a compact microchannel heat sink system to have lower thermal resistance and to work under high cooling load situations. The microchannel heat sink technology is therefore increasingly being used in modern electronic packaging, high-speed computers and other related industries. The heat exchanger design of the thermo-electric cooling module attempts to harness possible heat transfer enhancement in flow through narrow passages.

[0094] The preferred heat exchanger is made of a metal material having a high thermal conductivity and has several narrow rectangular passages through which the cooling liquid flows. High thermal conductivity helps to spread heat flux evenly around the channel walls that are in contact with the liquid, thereby increasing the effective area heat transfer to the fluid. Due to special flow characteristics in narrow passages as in microchannels, high heat transfer rates are present in the flow. The developing nature of the flow through the passage further contributes to the heat transfer augmentation. The combined effect of all these mechanisms gives rise to significantly low thermal resistance between the thermo-electric module attached to

the heat exchanger and the cooling fluid than previous designs of heat exchangers for similar applications.

[0095] Improvement of heat transfer to and from the surface of a thermoelectric module is of substantial commercial value. The coefficient of performance (COP) of a thermoelectric module is closely related to the temperature difference across it from the hot side to the cold side. COP is a measure of a heat pumps efficiency in transferring heat from one region to another, or in other words from a heat source to a heat sink, and is defined as the amount of heat moved from the source to the sink (in W) divided by the amount of power required (in W) to effect this movement. The present invention is aimed at reducing the thermal resistance at the surface of the module thus reducing the difference in temperature between the module and heat sink and source temperatures.

[0096] For example, if two thermoelectric modules are pumping 50 to 60 watts of heat (a typical heat load for a refrigerator) then a thermal resistance of 0.2°C/W will result in a temperature differential across the cold side of 5°C (25W x 0.2°C/W). That is, the internal temperature of the module must be 5°C colder than the desired temperature inside the refrigerator (usually 5°C). Thus the cold side of the module will be at 5 – 5 = 0°C

[0097] Because the electric power consumed by the module must be dissipated along with the heat removed, the hot side of each module will need to transfer at least 50 watts of heat (assuming the COP of the module is 1.0). The temperature difference between the internal hot surface of the module and the ambient will then be 10°C (50 x 0.2) if the thermal resistance is 0.2°C/W. At standard testing conditions the ambient temperature is 25°C therefore the hot side of the module will be at 25 + 10 = 35°C.

[0098] Thus the thermoelectric module will operate at a temperature differential of 35 - 0 = 35°C.

[0099] The thermal resistance of 0.2°C/W is typical of standard air cooled finned heat exchangers clamped to thermoelectric modules. A high efficiency conventional thermoelectric

heat exchanger, of the prior art, will give a thermal resistance of 0.1°C/W. By the same methodology as above the temperature differential will be 27.5°C.

[0100] According to the heat exchanger technology of the present invention, the thermal resistance is only of the order of 0.01 to 0.02°C/W. Using the same methodology as above, the thermoelectric module will operate at a temperature differential of only 20 + 1.8 = 21.8°C.

[0101] The increase in temperature differential across the module has the effect of reducing the COP. This does not affect the temperature differential on the cold side because the heat load remains constant and therefore the heat pumped through the cold side remains constant. However, because of the falling COP and therefore increased electrical power required, the amount of heat pumped through the hot side must increase. This compounding effect increases the temperature difference across the hot side and therefore across the module. Given the operating characteristics of the module an iterative style calculation is required to provide a solution.

[0102] The difference in COP for a theoretical 4cm x 4cm thermoelectric module operating under 25W heat load at three different thermal resistances is shown in Table 1 below. In the case of a thermal resistance of 0.2°C/W the dT is larger than that calculated above, demonstrating the detrimental affect of the reduced COP.

Thermal Resistance (°C/W)	0.2	0.1	0.02
dT	36.5	27.0	21.3
Optimum COP at dT	0.77	1.26	1.77

Table 1

[0103] The beneficial effect of reducing thermal resistance is clearly demonstrated. A thermal resistance of 0.02 means that the system will consume only 43% of the power that it will with a thermal resistance of 0.2°C/W.

[0104] Values below 0.1°C/W are uncommon and require special design consideration to be achieved.

[0105] A further significant feature of the heat transfer capabilities of the heat exchangers of the invention relates to the field of thin film and high heat flux thermoelectrics. These thermoelectric materials have substantially higher ZT levels than conventional bulk materials.

[0106] ZT is a dimensionless measure of the efficiency of heat pumping and conventional bulk materials have a maximum value of 1.0. Recently film materials have been produced with ZT's of 2.4 (at the Research Triangle Institute, North Carolina, USA) and 2.0 at Massachusetts Institute of Technology.

[0107] However a limitation on the usage of these new thermoelectric materials is the fact that they have considerably higher heat fluxes (the amount of heat flowing through a given cross-sectional area) than conventional bulk materials. A conventional high heat pumping module such as the Drift 0.8 from Kryotherm has a heat flux of 11 W/cm^2 at a dT of 0°C. Thin film materials, such as those produced at the Research Triangle Institute, have a heat flux up to 700 W/cm^2, a factor of 60 times higher.

[0108] The ability of the heat exchanger attached to the thermoelectric module to actually transfer heat into and out of the module becomes critical as heat flux increases. If the heat exchanger cannot cope with the heat flux then the module temperature differential quickly rises to a point where the module loses the capacity to pump heat.

[0109] Table 2 below shows the impact of thermal resistance on a moderately high heat flux module. This module has the characteristics of the thermoelectric modules which are being developed and is assumed to have a ZT = 3.0 with a surface area of 4cm x 4cm.

Heat load (W)	70	70	70	70
Thermal Resistance (°C/W)	0.2	0.1	0.02	0.01
dT (°C)	66.4	37.7	23.2	21.6
COP	0.76	1.91	3.56	3.89

Table 2

[0110] The extra value of thermal resistance (0.01°C/W) is included because improvements to this level are likely to be necessary. At this higher heat flux the impact of thermal resistance on COP is much greater.

[0111] At high heat fluxes, such as 200W per module, a thermal resistance of 0.2 or 0.1 will prevent the module from operating. The higher dT required across the module in order to pump the heat exceeds the capability of the material.

[0112] The improvement in COP for lower thermal resistances at high heat loads is shown below in Table 3. The module is the same as for Table 2 but with a higher heat load.

Heat load (W)	200	200	200
Thermal Resistance (°C/W)	0.05	0.02	0.01
dT (°C)	47.3	29.5	24.6
COP	1.37	2.64	3.32

Table 3

[0113] High heat fluxes in thermoelectric modules also lead to increased internal operating temperatures unless the heat can be removed by the heat exchangers. These increased temperatures translate into higher thermal stresses which can destroy the structural integrity of the module.

[0114] The lower thermal resistance of the heat exchanger's of the invention (by a factor of at least 10 over other commercial systems) means that ten times the heat flux can be transferred to a cooling fluid for the same temperature difference, allowing thin film and high heat flux thermoelectrics to operate at their optimum conditions with improved reliability.

[0115] The earlier embodiments emphasize the importance of increasing surface area to enhance the total amount of heat transfer. Relatively limited geometries were presented as suitable and of providing a low thermal resistance (less than 0.1°C/W).

[0116] Further analysis and test work has been carried out to evaluate a wider range of geometries and to consider the effect of variations in channel wall thickness as well as the fluid channel width and height. Heat flow is maximized when the temperature of the heat exchanger surfaces surrounding the fluid is even. The design of an optimized heat exchanger requires careful specification of channel height and width. In the analysis carried out below channel wall thickness is also considered as a third variable.

[0117] The advantage of a high heat transfer coefficient under laminar flow conditions is of lesser importance as the range of applications is broadened outside of domestic refrigeration and as the heat flux levels increase. A larger proportion of ancillary power can be used to pump heat exchange fluids around the system and this enables flow regimes to operate in the turbulent region. Heat transfer coefficients are significantly higher in turbulent flow than laminar flow.

[0118] As channel wall thickness increases, there is a reduction in thermal conduction resistance through the cross section. However, for a fixed heat exchanger face width, the number of channels must reduce if channel width remains constant. There is, therefore, a consequential loss of metal surface area engaged in heat transfer to the fluid. The increase in wall cross-sectional area versus the reduced channel surface area requires an optimized solution, which also must consider channel height.

[0119] The relationships between channel width, height, and wall thickness are complex. Regions of optimal performance can be shown on graphs of thermal resistance against increasing wall thickness (or decreasing (N)umber of channels) for a range of channel heights.

[0120] These relationships can more easily be represented as a graph of thermal resistance against the number of channels per meter of heat exchanger width for a given channel

width. Although a typical thermoelectric module and its heat exchanger is only about 4cm wide, expressing the number of channels per meter generalizes this parameter.

[0121] Narrowing channels reduces thermal resistance, but also increases pump power if channel height and the number of channels are kept constant. Increasing channel height increases the cross sectional area for fluid flow, reducing pump power required, and also increases surface area. But the benefit of channel height increase is lost above 6 to 10 mm because of thermal conduction losses in the extended channel wall. The extent of this thermal conduction loss also depends on the wall thickness as it governs the wall cross-sectional area in the direction of heat flow.

[0122] Fig. 25 shows a typical relationship between thermal resistance and channel height for various channel widths. Thermal resistance increases substantially if the channel height is reduced below a certain level. The loss of surface area overshadows reduced conduction losses as heat travels along the wall. Narrower channels exhibit reduced thermal resistance in all cases with different optimum heights depending on width.

[0123] The size of the heat exchanger is still relatively small at a channel height of 6mm. Where the heat exchangers are fixed to thermoelectric modules and used for a refrigeration application, which includes freezing, the flow rates and pumping power required are critically important. Liquid flow through the heat exchangers is in the laminar region to minimize pumping power. This is especially important in a freezing application where the liquid used will be at a temperature in the region of –20°C and therefore of a much higher viscosity than normal.

[0124] Flow rates generally used are from 0 liter per minute (lpm) up to 2.0 lpm per cm of width of the thermoelectric module. For a 4 cm wide thermoelectric module at 0.5 lpm per cm this equates to 2 lpm total flow rate. Flow under these conditions is therefore in the laminar region.

[0125] Where an application involves thermoelectric modules with high heat pumping capacity, such as in air conditioning, then the power consumed for ancillary purposes is less important. Flow rates can then be designed in the turbulent region.

[0126] Fig. 26 shows typical curves of Thermal Resistance (°C/W) on the Y axis against Number of channels/meter (N) for a range of channel widths. As shown, the channel height is a constant 6 mm. In the case of the exemplary graphs below, the fluid is water at 27°C. Other liquids can be used, however, the optimum dimensions remain substantially the same. All graphs relate to an exemplary 4cm x 4cm size heat exchanger.

[0127] Graphing thermal resistance against the number of channels per meter of heat exchanger width is a convenient method of evaluating wall thickness as a geometrical constraint. It has the advantage of returning the minimum number of channels of a certain width necessary to ensure a thermal resistance of less than 0.1°C/W. This then defines the wall thickness allowable. Fig. 26 shows this relationship for a range of channel widths from 0.1 mm to 1.0 mm.

[0128] A thermal resistance of less than 0.1°C/W is desirable where thermoelectric applications are sensitive to efficiency.

[0129] From the graphs shown in Fig. 26, we can select a minimum allowable number of, for example, channels of 200 per meter for the 1.0 mm wide channels and 25 per meter for the 0.1 mm wide channels can be selected. Table 4 shows the minimum number of channels per meter for various channel widths is shown below. Fig. 27 shows this data plotted graphically.

Channel width (mm)	Min No. Channels/m
0.05	25
0.1	25
0.2	55
0.3	80
0.4	110
0.5	145
0.6	170
0.7	200
1.0	310

Table 4

A very good linear approximation can be expressed by the formula:

$N = 314*W - 13$ where N = minimum number of channels/meter

W = channel width (mm)

[0130] Very narrow channels, e.g. 0.05 to 0.3 mm, exhibit lower thermal resistance across the complete range than do wider channels. For example, a heat exchanger with 200 channels per meter at 0.1 mm width has about one quarter the thermal resistance of a similar heat exchanger with 200 channels per meter at 1.0 mm width.

[0131] All channel width curves (0.05 mm to 1.0 mm) show a reduction in thermal resistance with increasing number of channels (or decreasing wall thickness). However, even though increasing the number of channels increases the surface area available for heat transfer there is a minimum wall thickness which optimizes thermal resistance. Below this wall thickness the overall thermal resistance rises due to conduction resistance to heat flow up the wall becoming the predominant factor.

[0132] This is shown in the Fig. 28 below which plots thermal resistance against wall thickness for channels from 0.05 mm width to 1.0 mm width.

[0133] As can be seen, this minimum wall thickness is less than 0.5 mm. From the curves in Figure 28 for a 1.0 mm channel width has an optimum wall thickness minimum of 0.5 mm. For a 0.05 channel width the optimum wall thickness is 0.05mm and for a 0.1mm channel width the optimum wall thickness is 0.1mm. The range of wall thicknesses of value to the design of the invention starts from below this optimum thickness (say 50%). Below this starting value the thermal resistance rises unacceptably.

[0134] Table 5 and Fig. 29 below show the values of minimum wall thickness for the range of channel widths:

Channel width (mm)	Min wall thickness(mm)
0.05	0.025
0.1	0.025
0.2	0.05
0.3	0.10
0.4	0.15
0.5	0.175
0.6	0.20
0.7	0.235
1.0	0.25

Table 5

As channel width increases there are fewer channels and less surface area exposed for heat transfer to the fluid. The form of the curve shows that the ratio of wall thickness to channel width reduces in order to compensate for the reduced surface area. Eventually the minimum wall thickness plateaus at 0.25mm. A good approximation for the minimum wall thickness is given by the formula:

$M = 0.6*W - 0.3*W^2$ where M = minimum wall thickness (mm)

W = channel width (mm)

[0135] In actual practice, channel widths may be determined by other factors apart from thermal resistance, such as liability to fouling and blockage and available pump power. It is therefore very important to be able to optimize the channel wall thickness and height once the channel width is fixed and the optimal ranges presented below enable us to do this.

[0136] Channel dimensions are shown in Fig. 30 where:

T = wall thickness

H = channel height

W = channel width

[0137] From the graphs showing the relationship of thermal resistance to number of channels per meter and minimum wall thickness a range of acceptable geometries can be chosen for each value of channel width to ensure that thermal resistance is below 0.1 °C/W for a 40 mm heat exchanger width. The ranges are shown below in Table 6.

Channel width *W* (mm)	Min No. Channels/m	Min wall thickness *T* (mm)
0.05	25	0.025
0.1	25	0.025
0.2	55	0.05
0.3	80	0.10
0.4	110	0.15
0.5	145	0.175
0.6	170	0.20
0.7	200	0.235
1.0	310	0.25

Table 6

These relationships can be expressed numerically by the formulae:

$$N = 314*W - 13 \quad \text{where } N = \text{minimum number of channels/meter}$$

W = channel width, and

$$M = 0.6*W - 0.3*W^2 \quad \text{where } M = \text{minimum wall thickness (mm)}$$

W = channel width (mm)

These optimized dimensions are valid for all channel heights between 0 mm and 10 mm

[0138] Cross sectional views of some typical and useful actual heat exchangers are shown in Figs. 31 to 33. The heat flows from the base of the heat exchanger through the walls and up to the top plate. Materials of construction are preferably highly thermally conductive. The heat exchanger can be manufactured wholly of parent metal by an extrusion process or may be made up of separate parts. Where it is made from separate parts they are joined by methods which provide low thermal resistance, such as welding or soldering. Where channel heights are relatively small, a chemical etching process can be utilized to form channels.

[0139] The heat exchanger 300 shown in Fig. 31 has a base 301, a top plate 302 and spaced apart walls 303 extending between and being integral with the base 301 and the top plate 302. Such a heat exchanger 300 can be made by an extrusion process.

[0140] The heat exchanger 400 shown in Fig. 32 has a base 401, a top plate 402 and ends 403. The internal walls 404 are formed from a strip of folded metal, there being a solder joint between the lower parts 405 of the walls 404 and the base 401 and between the upper parts 406 of the walls 404 and the top plate 402.

[0141] The heat exchanger 500 shown in Fig. 33 is similar to that of Fig. 32 with the internal walls 505 being formed from compressed metal sheet, ends 503, and there being a solder joint between the lower part 505 of the walls 504 and the base 501 and between the upper parts 506 of the walls 504 and the top plate 502.

[0142] While the invention has been particularly shown and described with reference to preferred embodiments thereof, it will be understood by those skilled in the art that various alterations in form and detail may be made therein without departing from the spirit and scope of

the invention. In particular, the specific shape and size of the heat exchanger, the shape and specific number of the pieces, the arrangement of the sections, and the means for installing the various sections can be altered depending on the specific application without departing from the scope of the invention.